

# Vibration Characteristics of Stirling Cycle Cryocoolers for Space Application

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## ABSTRACT

Vibration generated by small Stirling-cycle cryocoolers is an important concern for spacecraft designers planning to incorporate these cryocoolers into near-term space-science instruments. Under joint Air Force/Ballistic Missile Defense Organization (BMDO) and NASA Eos/AIRS Instrument sponsorship, JPL has an extensive ongoing cryocooler characterization program addressed to measuring the important cryocooler performance characteristics such as self-generated vibration; these measurements provide the interface and trend data needed by instrument designers as well as valuable feedback for improving cryocooler performance.

Self-generated vibration has been measured at JPL for a wide variety of Stirling cryocoolers, from early engineering development model coolers to flight coolers. These data, when viewed as a whole, provide insight into a variety of generic cryocooler vibration trends. Data are presented which characterize the sensitivity of cryocooler vibration to operational parameters such as stroke and drive frequency, to different harmonic cancellation approaches, and to the influence of the stiffness of the cooler structural mount. The data are presented for all three translational axes as well as for the moment about the drive axis.

## INTRODUCTION

As part of JPL's overall cryocooler characterization program sponsored jointly by the Air Force/BMDO and the NASA Earth Observing System AIRS Instrument Project, a variety of state-of-the-art space Stirling cryocoolers have been tested for their vibration attributes.<sup>1,2</sup> This paper discusses the vibration behavior of several Stirling-cycle cryocoolers designed for long-life spacecraft cooling in the 50 to 150 K temperature range. All incorporate flexure springs and clearance seals in the Oxford-cooler tradition to prevent wear and prolong life. This feature separates them from the less expensive tactical Stirling cryocoolers that use piston rubbing as the means of guiding the piston and displacer in their cylinders. The tested coolers are designed to use identical back-to-back compressor or displacer pairs, or an active mass counterbalancer, to achieve momentum compensation in the active piston/displacer drive direction. All are designed to be powered by sophisticated drive electronics incorporating active vibration control algorithms based on force, acceleration, or piston position feedback; none provide active control over vibration normal to the piston/displacer drive axis.

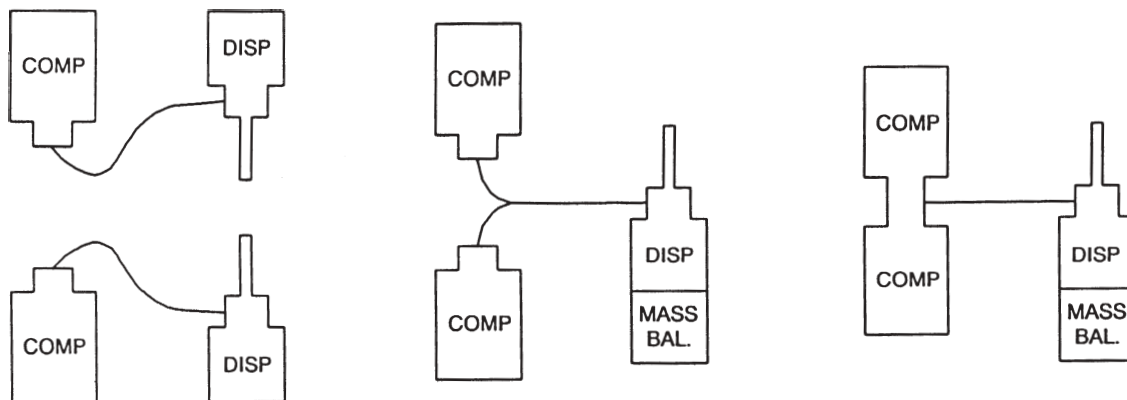


Figure 1. Three principal back-to-back cooler configurations.

Momentum compensated cryocooler configurations fall into three generic classes shown schematically in Fig. 1. These include (1) two separate coolers (consisting of a single compressor and single displacer) mounted back-to-back, (2) a cooler having two separate back-to-back compressors manifolded into a single helium transfer line feeding a single displacer, and (3) a single cooler having two back-to-back compressor pistons sharing a common compression space connected to a single displacer. The latter two configurations use an active mass balancer to suppress the vibration of the single displacer. Tested coolers that fall into these categories include: (1) the BAe 80K and 50-80K coolers and the Lockheed/Lucas 65K SCRS cryocooler, (2) the Stirling Technology Company (STC) 80K TDM cryocooler, and (3) the Hughes 65K SSC cryocooler and the Lockheed/Lucas 1710C cryocooler, respectively.

## MEASUREMENT APPROACH

In characterizing cooler-generated vibration it is useful to speak in terms of the peak vibratory force imparted by the cooler into its supports when rigidly mounted. This force is the reaction force to moving masses within the cooler that undergo peak accelerations during various phases of the cooler's operational cycle. The accelerations can be from controlled motion such as the reciprocating sinusoidal motion of the Stirling compressor piston and displacer, or from natural vibratory resonances of the cooler's elastic structural elements. Sensitive spacecraft instruments dictate that such force levels be on the order of 0.2 N or less in any direction.

Compressor and displacer resonance and vibration characterization measurements are made with the cooler mounted in JPL's specially constructed six-degree-of-freedom force dynamometer. The dynamometer is instrumented with six load cells whose amplified output signals are recorded in real time using a spectrum analyzer. The cooler component not currently under characterization is solidly mounted to an adjacent rigid structure to minimize transmission of vibration from this component to the component under test. The JPL dynamometer has a useful frequency range from 10 Hz to 500 Hz and a force sensitivity from 0.005 N to 445 N full scale. Passive viscoelastic damping techniques are employed to minimize coupling of the cooler vibration into the dynamometer support structure resonances during the measurements.

When discussing cooler vibration sensitivities it is useful to divide the generated vibration into at least four categories: 1) the vibration in the piston drive direction (Z-axis for the JPL dynamometer) at the fundamental drive frequency of the cooler, 2) vibration in the Z-axis drive direction at upper harmonics of the fundamental drive frequency, 3) vibration in the lateral, cross-axis directions (X and Y axes), which are perpendicular to the drive axis, and 4) oscillatory twisting about the spindle axis caused by the flexure springs winding and unwinding as they are stroked longitudinally.

The vibration characterization results presented in the remainder of this paper are displayed in the form of spectral plots of force in each of the three principal axes, and moment about the



spindle-stroke (Z) axis. Most plots are parameterized with respect to a variable of interest to illuminate a particular parameter sensitivity. Sensitivity studies are presented for compressor and displacer stroke amplitude, perturbation of the drive frequency and piston offset, cryogenic versus room-ambient coldtip temperature, and degree of back-to-back and active harmonic cancellation. The individual plots have been chosen as representative of generic trends observed with many coolers, and are drawn from over 3000 spectra gathered from 13 coolers.

## **VIBRATION CHARACTERISTICS OF A SINGLE COMPRESSOR**

### **Stroke Sensitivity**

Without momentum cancellation, the vibration of a single cooler along the drive axis (Z-axis) at the fundamental drive frequency is linearly proportional to the maximum piston acceleration and the moving mass. For sinusoidal motion, it is therefore linearly proportional to the piston drive amplitude and to the square of the drive frequency. This classical behavior is seen in the data for the BAe 80K and 50-80K coolers as shown in Figs. 2 and 3.

In contrast to the fundamental vibration in the drive axis, vibration at higher harmonics and in the cross axes is caused by subtle non-linear attributes of the cooler. The parameters thought to be primarily responsible for the higher order and cross-axis vibration harmonics include nonlinearities in the drive motor and spring stiffnesses as a function of stroke, and the stroke-dependent internal resonant frequencies coinciding with a harmonic of the drive frequency. As various structural resonance modes get excited at different stroke amplitudes and frequencies, vibration levels can change abruptly. Note the order of magnitude changes in the high-order harmonics of the BAe coolers in Figs. 2 and 3 as the compressor stroke amplitude is changed in 1-mm increments. This is typical for coolers of this type. Note also that the fundamental shape of the vibration spectra is similar for the two BAe coolers, even though the piston diameters and moving masses are quite different. The spectra in later plots for other types of coolers will be seen to have different shapes.

### **Sensitivity to Stroke Offsets**

Figure 4 illustrates the sensitivity of the vibration spectra of the BAe 50-80K cooler to slight longitudinal offsets of the piston around its neutral spring position. Overall the effects are small, but close examination reveals order of magnitude changes for individual selected harmonics.

### **Sensitivity to Perturbations in the Drive Frequency**

The cooler drive frequency is generally selected to optimize the thermal performance of the cooler. While thermal performance changes very little over a narrow range of frequencies, the vibration levels caused by exciting internal and external structural resonances can vary significantly; these structural resonances often have important implications on the performance of the cooler in its host application. Figure 5 describes the BAe 50-80K compressor's vibration sensitivity to perturbation of the drive frequency by 1-Hz increments around its nominal value. As with the piston offset, the overall effects are small, but close examination reveals order of magnitude changes in individual selected harmonics.

## **VIBRATION NULLING WITH BACK-TO-BACK COMPRESSORS**

The most straightforward approach to suppressing cooler vibration is to mount two coolers back-to-back or together with an active balancer to provide momentum cancellation. Previous measurements have confirmed that this technique works reasonably well for the first two harmonics, but can leave modest degrees of residual vibration in the cross axes and upper harmonics.<sup>1,3</sup> Figures 6 and 7 illustrate the residual vibration measured for the Stirling Technology 80K cooler and the Lockheed-Lucas SCRS cooler. The residual vibration for these coolers is quite similar to earlier data on back-to-back BAe 80K coolers.<sup>1</sup> Note that the fundamental drive-axis vibration has been nulled to a high degree as would be required for a

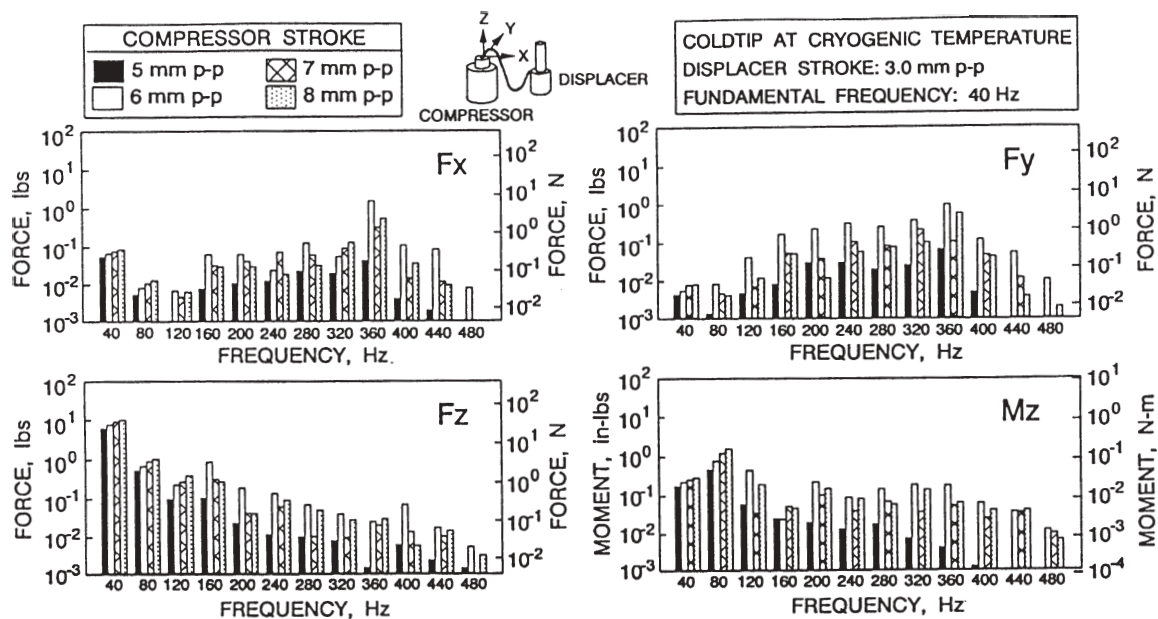


Figure 2. Compressor vibration spectra for the British Aerospace 80K cooler as a function of compressor stroke.

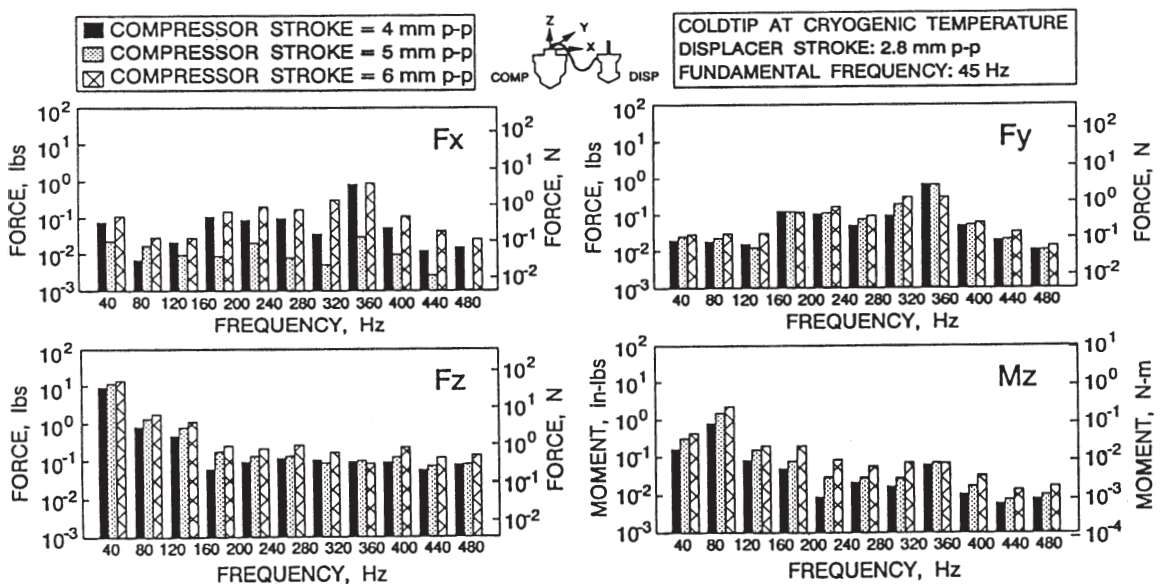


Figure 3. Compressor vibration spectra for the British Aerospace 50-80K cooler as a function of compressor stroke.

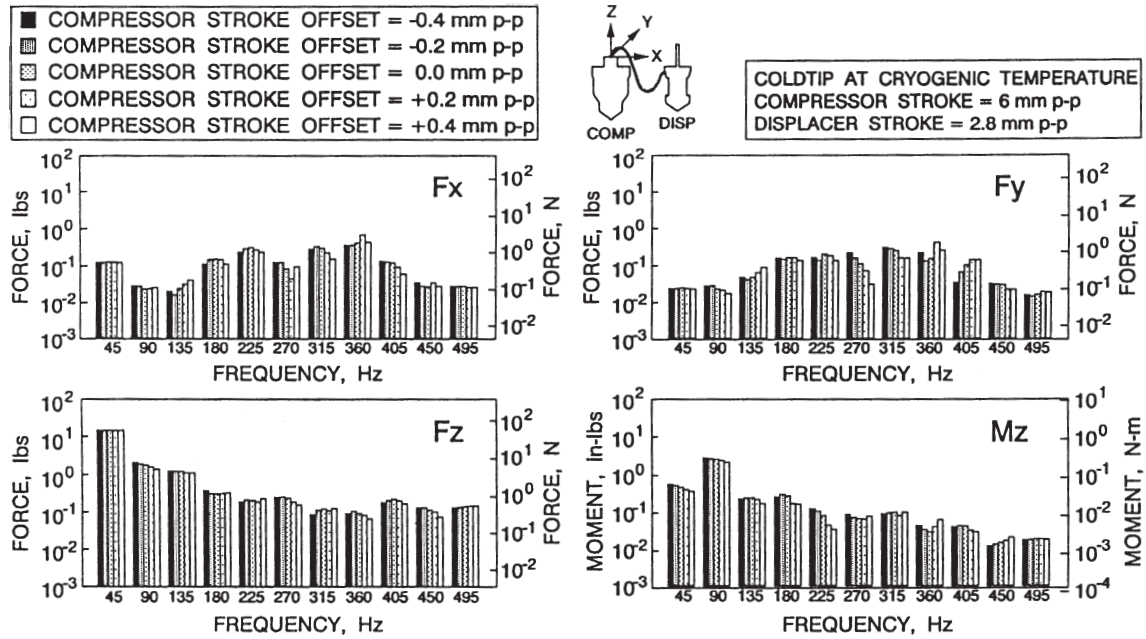


Figure 4. Compressor vibration sensitivity of the British Aerospace 50 - 80K cooler as a function of stroke offset.

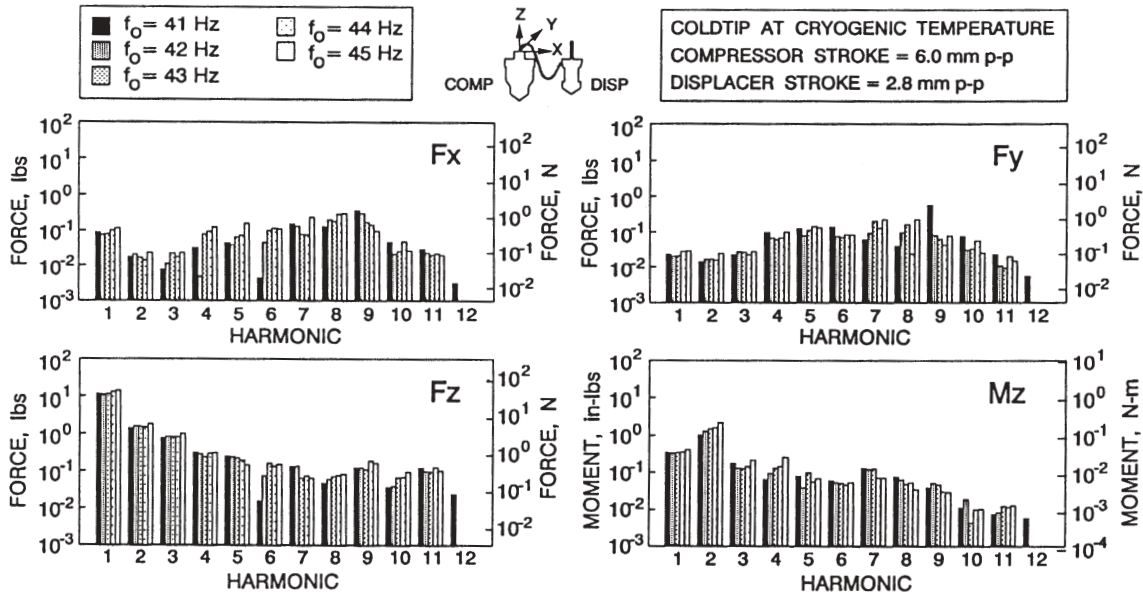


Figure 5. Compressor vibration sensitivity of the British Aerospace 50 - 80K cooler as a function of drive frequency.



sensitive space-instrument application; however, considerable upper harmonic vibration remains. Also note the strong sensitivity of the residual vibration to the stroke amplitude; this follows the trends seen in Figs. 2 and 3 for the BAe coolers.

### **Sensitivity to Cold-tip Temperature**

Because a cooler may operate at different cold-tip temperatures in different applications, it is important to understand the manner in which the generated vibration changes with cold-tip temperature. Figure 8 describes the minimal change commonly seen over the range of temperatures from room temperature down to 60K and below; i.e., there is no strong temperature effect.

### **Sensitivity to Perturbations in the Drive Frequency**

Because of the localized sensitivity to drive frequency noted above, it is useful to examine the effect of small perturbations of the drive frequency on the ability of compressor halves to null one another. Figure 9 describes the effect of 2-Hz frequency perturbations about the nominal 56-Hz drive frequency of the Lockheed SCRS cooler. The effect is measurable, but overall rather modest. This is typical for most coolers measured.

### **DISPLACER VIBRATION, SINGLE AND BACK-TO-BACK**

In general, vibration sensitivity of a the Stirling displacer is found to closely parallel the trends found for the compressor. However, the vibration force levels are generally lower due to the smaller moving mass. Figures 10 and 11 illustrate the vibration generated by the single displacer of a BAe 80K cooler and the back-to-back pair of displacers of the Lockheed-Lucas SCRS cooler. Note that the displacer vibration levels are significantly lower than those for the compressors, and that the displacer vibration is modestly sensitive to the compressor stroke.

### **VIBRATION SUPPRESSION WITH ACTIVE HARMONIC NULLING**

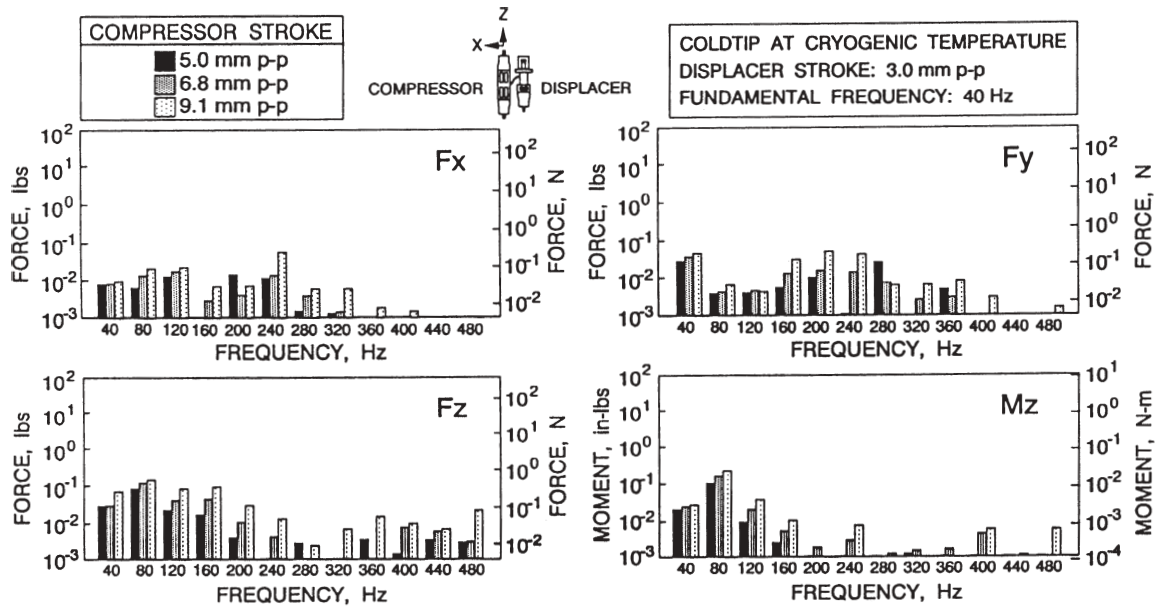
Over the past few years considerable progress has been made in the development of vibration control drive electronics designed to suppress the large vibrational forces of the first few harmonics in the axial drive direction.<sup>4,5</sup> Although, vibration suppression on the order of 20 to 40 dB has been repeatedly demonstrated, suppression of the axial vibration has been found to have little or no effect on canceling radial vibration components. Examples of the level of vibration control using force feed-back is shown in Fig. 12 for the STC cooler, and in Fig. 13 for the Hughes 65 K SSC cooler.

### **EFFECT OF COOLER INTERFACE MOUNT RIGIDITY**

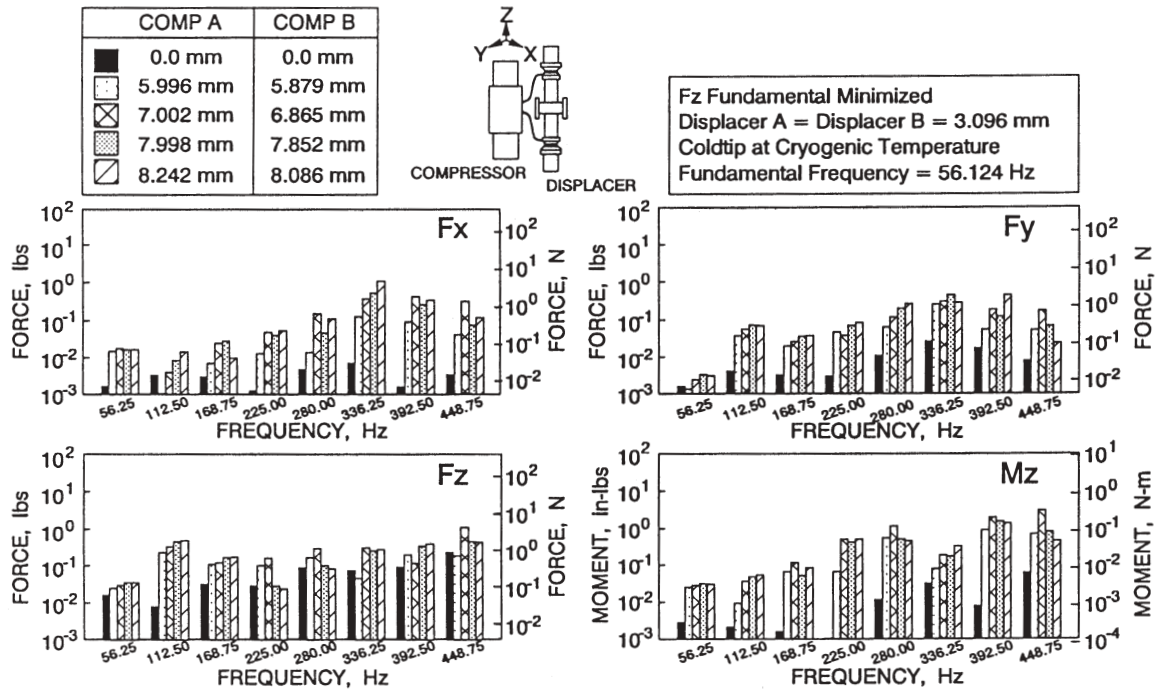
Cryocooler vibration control drive electronics work best under conditions where structural resonances of the cooler support lie well away from the 10 to 500 Hz frequency range where the key cooler generated vibrational harmonics occur. To avoid configuration specific results, measurement test setups such as the JPL dynamometer also try to keep the test system resonances outside of this frequency region.

However, when the cooler is integrated into a practical spacecraft instrument, the instrument will have numerous structural resonances in the 100- to 500-Hz range. It is likely that one or more of the cooler harmonics will coincide with these structural resonances, thus leading to possible control stability issues.

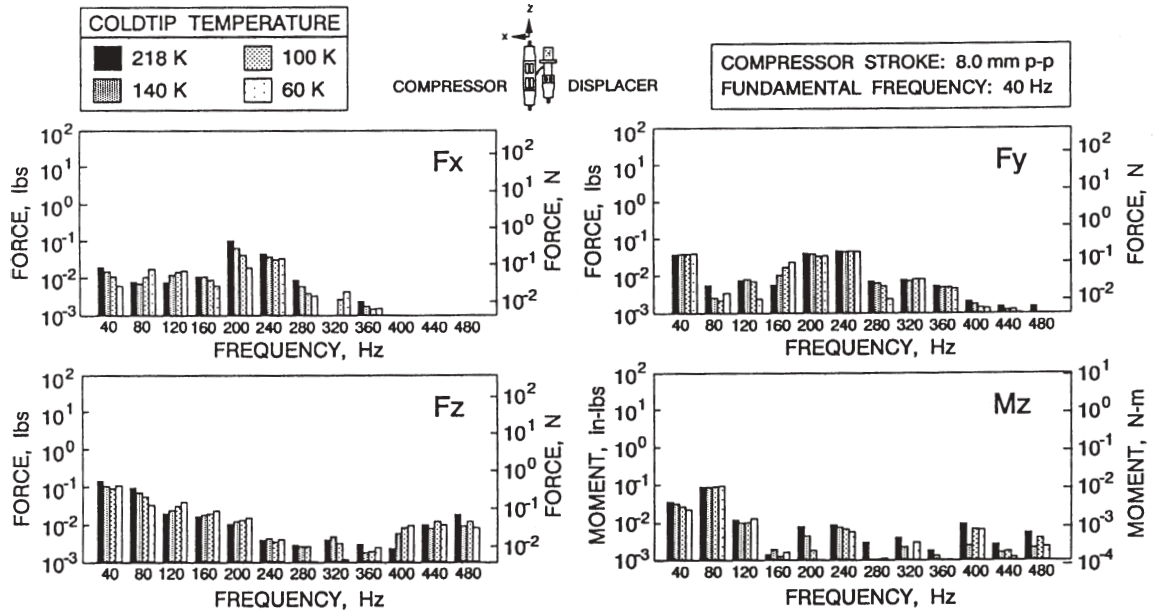
A second important ramification of the finite stiffness of the cooler structural interface is the likelihood of transmission of vibration between the compressor and displacer via the transfer line, thermal conductance straps, and integrating structures. With the displacer and compressor driven phase-synchronously at the same frequency, it is likely that the drive electronics will find it difficult to discriminate between the two units and maintain stable vibration control.



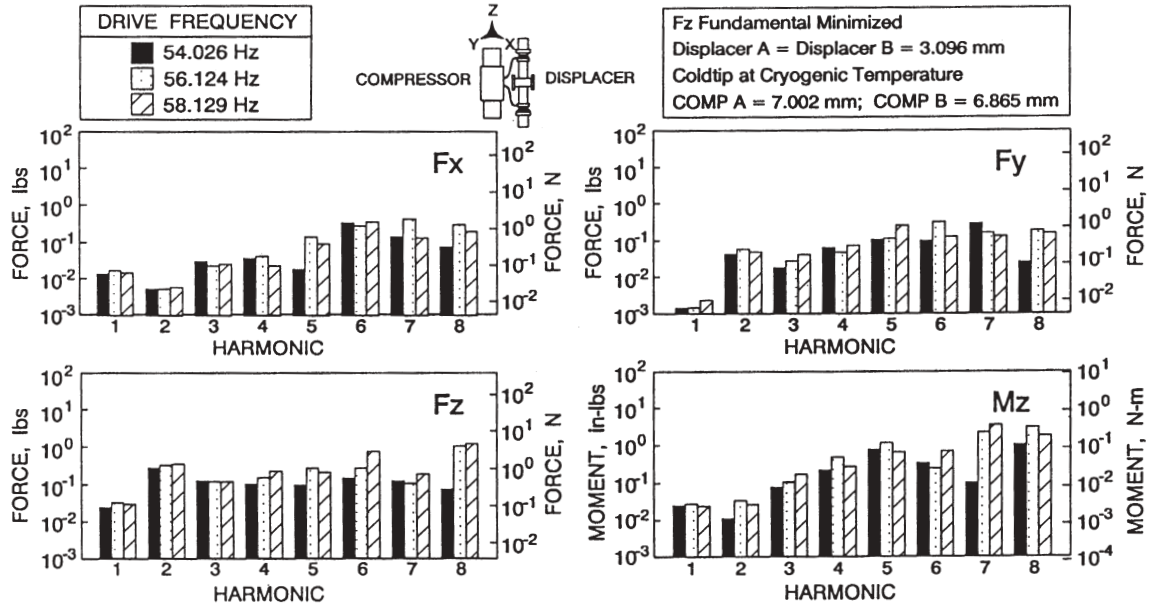
**Figure 6.** Vibration spectra for the back-to-back compressors of the Stirling Technology 80K cooler as a function of compressor stroke.



**Figure 7.** Vibration spectra for the back-to-back compressors of the Lockheed-Lucas SCRS cooler as a function of compressor stroke.



**Figure 8.** Vibration sensitivity to cold-tip temperature for the Stirling Technology 80K cooler.



**Figure 9.** Vibration sensitivity to small changes in the drive frequency for the Lockheed-Lucas SCRS cooler.



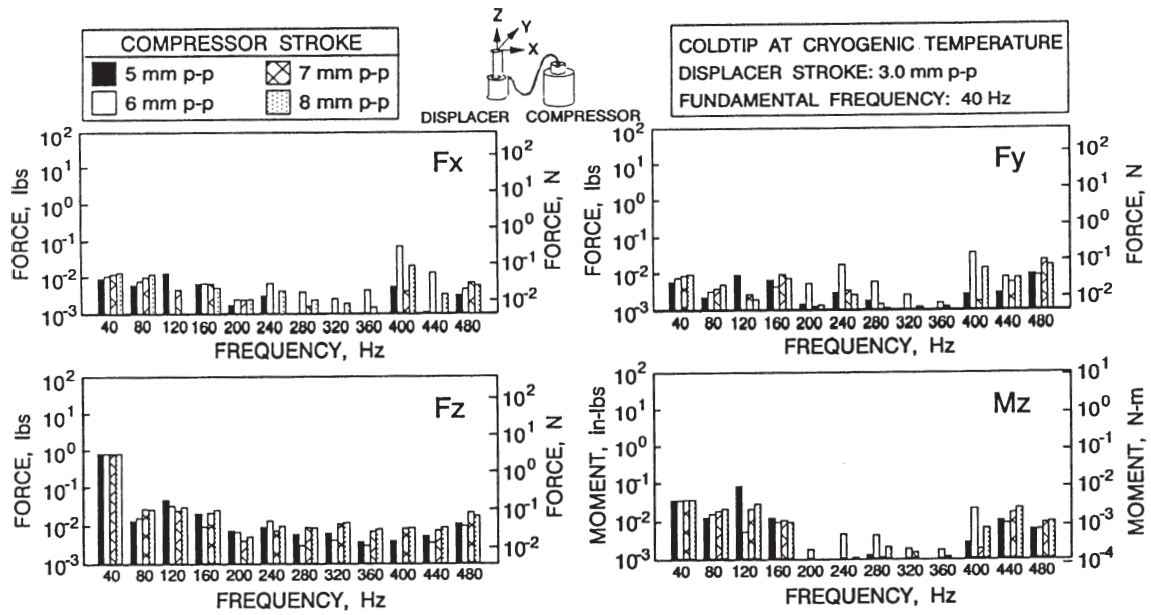


Figure 10. Displacer vibration spectra for the British Aerospace 80K cooler as a function of compressor stroke.

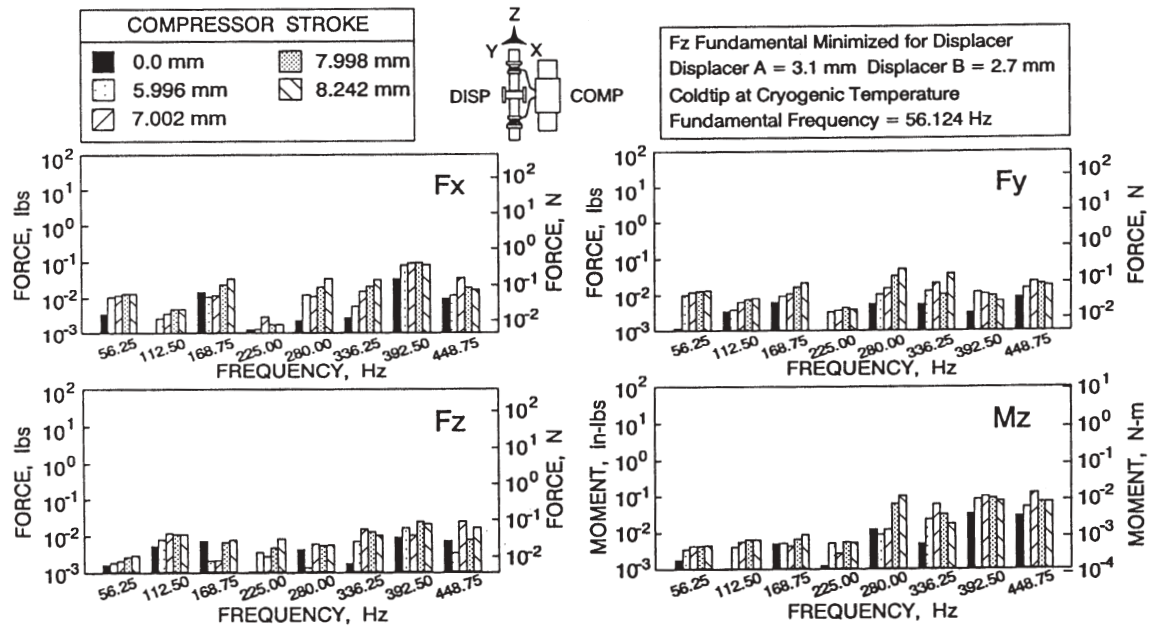


Figure 11. Vibration spectra of the back-to-back displacers of the Lockheed-Lucas SCRS cooler as a function of compressor stroke.

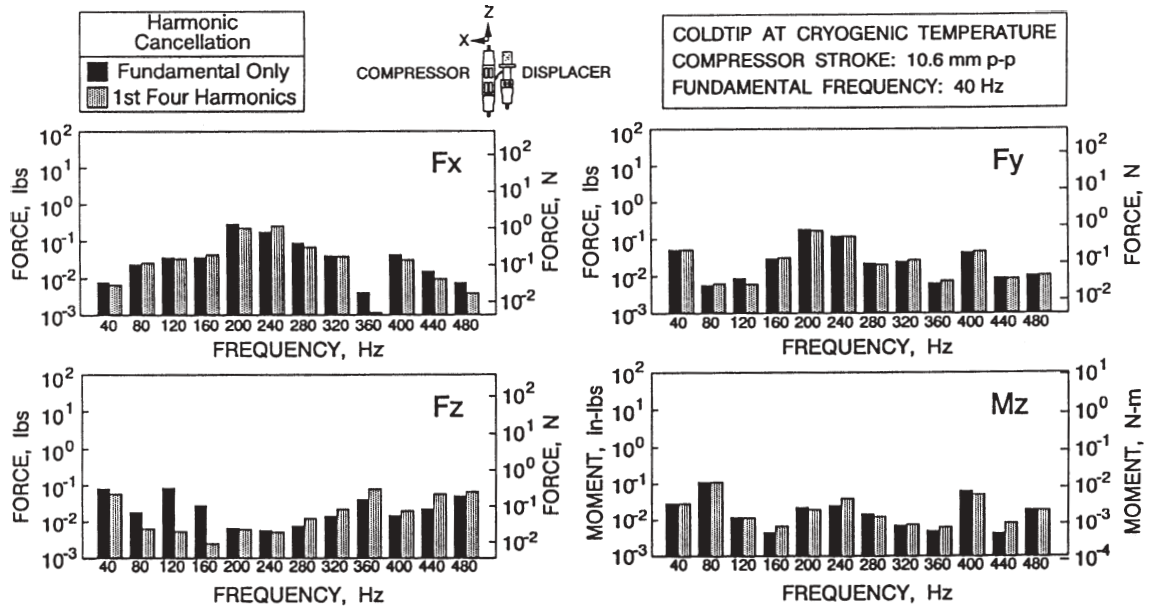


Figure 12. Vibration spectra for the back-to-back compressors of the Stirling Technology 80K cooler for two levels of harmonic cancellation.

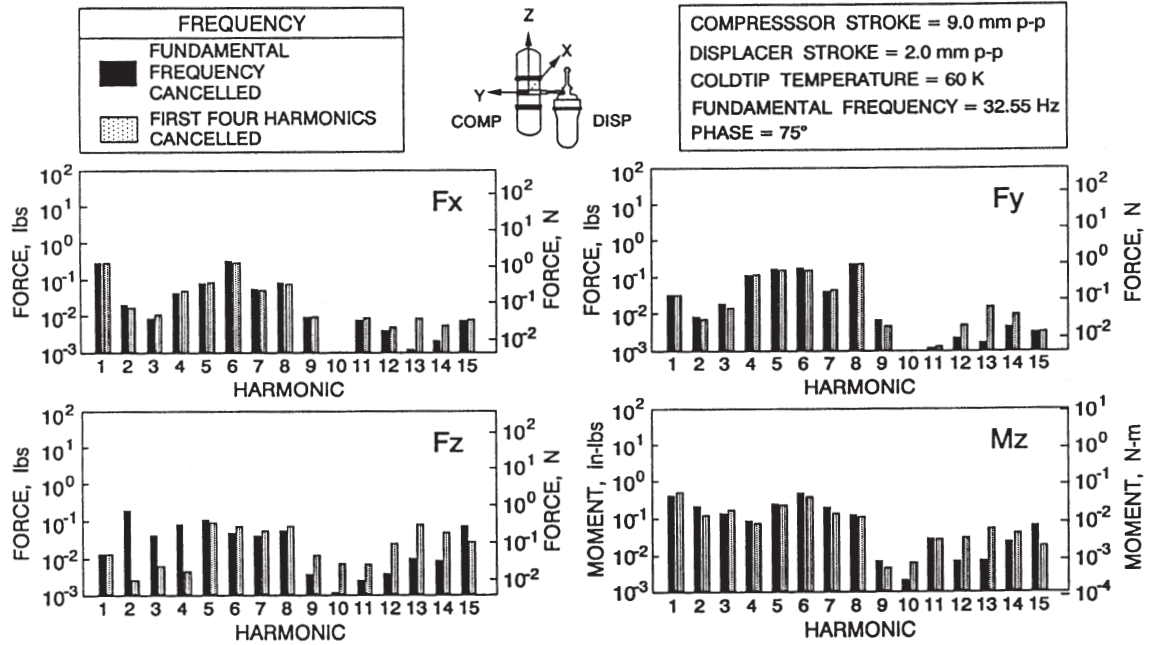
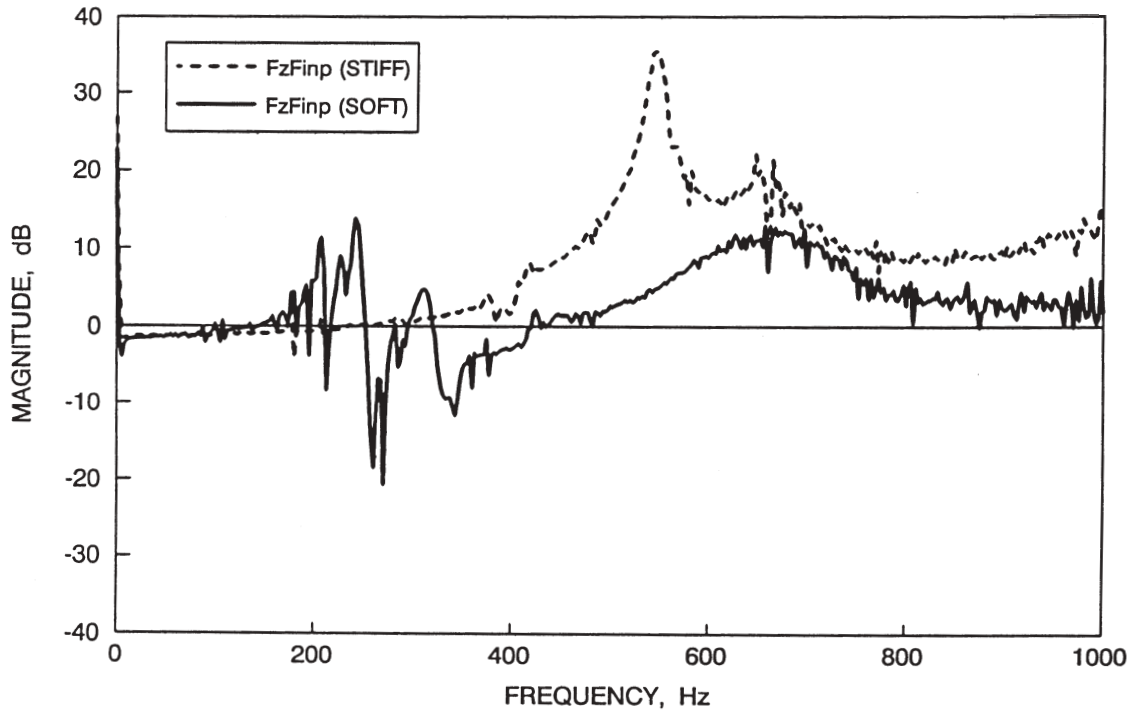


Figure 13. Vibration spectra for the back-to-back compressors of the Hughes SSC 65K cooler for two levels of harmonic cancellation.



**Figure 14.** Comparison of the Z-axis resonant response of the JPL dynamometer with both rigid and soft mounts.

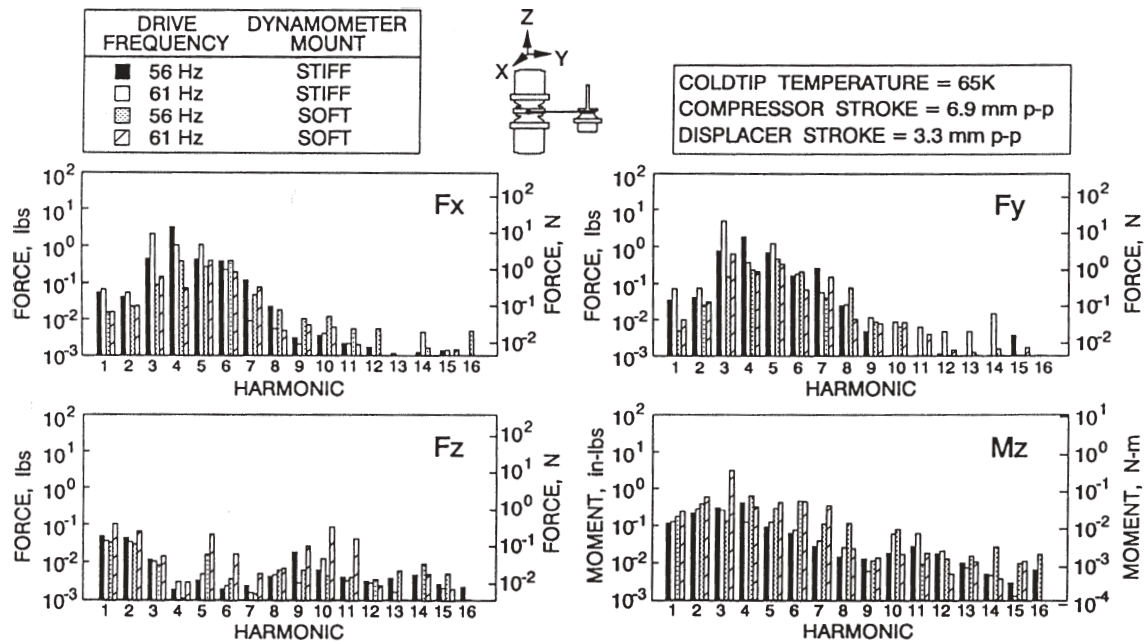
To address these questions, a series of tests were conducted at JPL with the dynamometer "soft-mounted" on springs as a means of simulating a practical spacecraft instrument structure. The soft mounted dynamometer provides the ability to determine the robustness of the cooler electronics vibration cancellation algorithms to flexible spacecraft modes in the frequency range of interest. The resonances of the soft dynamometer, noted in Fig. 14, can be adjusted in both amplitude and frequency by changing the stiffness of the springs, or by adding damping mass to the dynamometer structure. This allows a dynamometer resonance to be aligned with a drive harmonic of the cooler under test to simulate a worst-case operational condition.

A Lockheed-Lucas 1710 cooler having force feed-back vibration control was used in a series of pathfinder experiments. While on the rigidly-mounted dynamometer, the force feed-back loop's gain and phase were adjusted to provide optimum vibration suppression. Then spring mounts were installed under the dynamometer, and adjusted to give a desired stiffness. A dynamometer resonance was generated near 245 Hz, between the fourth and fifth harmonic of the cooler drive frequency. The cooler was able to operate in a stable condition. However, when the cooler drive frequency was adjusted to have the fourth harmonic coincide with the dynamometer resonance, the cooler drive electronics went unstable. The cooler could achieve stable operation once the gain and phase of the force feedback loops were readjusted to cancel out the fourth harmonic of the vibration, but at the expense of vibration suppression of the other harmonics. A comparison of the cooler vibration spectra for these conditions is shown in Fig. 15.

## SUMMARY AND CONCLUSIONS

Meeting the performance goals of near-term space-science instruments places demanding requirements on long-life space Stirling-cycle coolers. Achieving acceptably low levels of vibration from the cooler remains one of the most challenging. Development efforts with active vibration control electronics have demonstrated modest gains in reducing the cooler-generated axial vibration levels, but control stability remains an issue to be demonstrated. Vibration suppression of cooler-generated radial vibration is an area requiring much attention if overall cooler vibration levels are to be reduced to the 0.2-N level for spaceborne operation. The ability





**Figure 15.** Influence of dynamometer stiffness on the vibration suppression capability of the Lockheed-Lucas 1710 cooler.

to achieve stable vibration control of cooler operation over a range of drive frequencies necessary to minimize interaction with instrument resonances will greatly enhance the chance of successful operation in space.

## ACKNOWLEDGEMENTS

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